

Lowering Friction in Timing Chain Drive Systems by Tuning Tensioner Materials

Marcel Meuwissen
DSM Ahead B.V.

Jippe Van Ruiten
DSM Engineering Plastics

Thijs Besseling and Robbert van Sluijs
DSM Ahead B.V.

Maik Broda and Brian Pearce
Ford Motor Company

Fenton I. O'Shea
BorgWarner Morse TEC

ABSTRACT

Fuel economy improvement efforts in engines have focused on reducing parasitic losses. This paper addresses the friction losses in the valve train chain drive system where about half of the losses is caused by the chain sliding on plastic guide and tensioner arm faces (Figure 1). Efforts have been made to reduce these friction losses by optimizing the chain link profile, the geometry of the guide and tensioner arm rails, and developments towards low friction materials.

This paper describes the approach taken for the development of new low-friction chain tensioner arm plastic materials. The approach is characterized by building an understanding of the friction mechanisms and identifying the most critical material's properties. A lab-scale test is used for a first assessment of the friction performance of materials. The correlation between this lab-scale test and the actual chain-on-tensioner arm application is discussed. The effect of a number of key parameters such as temperature, oil viscosity, oil age, and surface roughness is illustrated and explained. Finally, the performance of a new low-friction polyamide 46 based material is demonstrated.

CITATION: Meuwissen, M., Ruiten, J., Besseling, T., van Sluijs, R. et al., "Lowering Friction in Timing Chain Drive Systems by Tuning Tensioner Materials," *SAE Int. J. Fuels Lubr.* 10(1):2017, doi:10.4271/2017-01-0462.

INTRODUCTION

One of the major challenges for the automotive industry is meeting emission legislative requirements. This is also driven by customer demands on reduced fuel consuming cars. The main concept to reduce fuel consumption is decreasing the overall weight of the car. The overall car weight is continuously increasing over the last years however. This results from increased safety and comfort demands.

The use of lighter materials is preventing an even stronger weight increase. Another route is the development of more efficient engines for reduced fuel consumption. One way to improve engine efficiency is the reduction of friction. Reducing mass of components can decrease the friction due to lower dynamical loads. This is mainly achieved by using lighter materials or better designs for moving components.

This paper focusses on reducing sliding friction within the valve train drive system by tuning the intrinsic tribological properties of plastics used for chain tensioner arms and guides. Within valve train drive systems, timing chains are used in roughly half the engines that come to the market in Europe. To maintain the chain tension throughout the engine's life, a tensioner arm is used at the slack side of the chain. In addition, guides and tensioner arms are used to prevent lateral resonance in the chain.

A breakdown of the friction losses in a typical timing chain system of an I3 or I4 engine is shown in Figure 1. Nearly 45% of the losses are attributed to chain to tensioner arm and guide contacts.

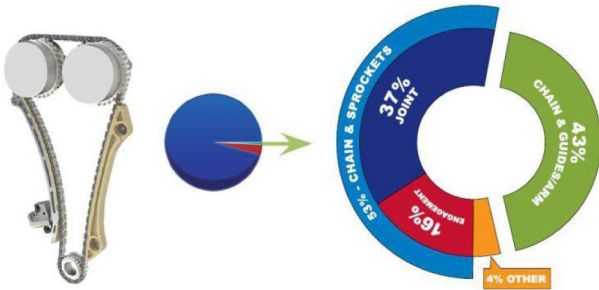


Figure 1. Breakdown of chain drive losses in typical I3 or I4 engines [1].

FRICITION IN A CHAIN-ON-GUIDE CONTACT

An example of a timing chain system is shown in Figure 2. This particular system incorporates one tensioner arm and three guides. A chain tensioner arm is used at the slack side of the chain to maintain the tension throughout the engine's life. Chain guides are installed between sprockets, to prevent lateral resonance in the chain parts.

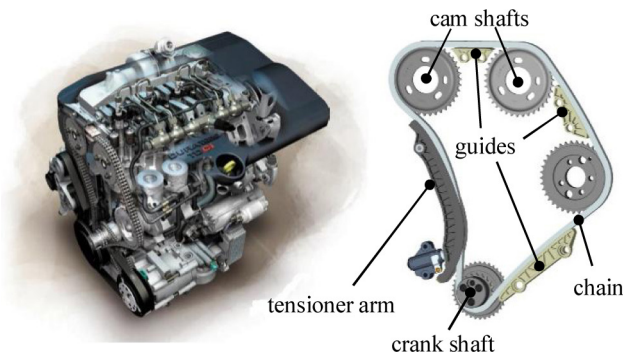


Figure 2. Example of a timing chain system on a Ford 2.4L Duratorq TDCi engine (left image) and sketch of the chain path, sprockets, tensioner arm, and guides (right image).

The tribology of these contacts is not only determined by the chain, tensioner arm, and guide surfaces, but also by the engine oil. Engine oil is sprayed onto the chain to reduce friction due to chain articulation and migrates to the chain to tensioner arm and guide contacts as well.

The friction in this lubricated contact depends on conditions such as chain tension, temperature, and sliding velocity. Friction under lubricated conditions is commonly represented by a Stribeck curve [2], which is schematically shown in Figure 3. On the vertical axis, the (dynamic) coefficient of friction (μ) is plotted, which is defined as the friction force at the contact divided by the normal force. On the horizontal axis the Hersey number is plotted. This number is defined as the product of sliding velocity (v) and oil viscosity (η) divided by normal load (p).

When the sliding velocity is low, the pressure in the oil film is unable to support the normal load induced by the chain force and asperities on both surfaces are in direct contact with each other. Friction characteristics in this regime are quite similar to those of friction under dry conditions, although friction levels are generally much lower due to the presence of a very thin layer of oil. This domain is termed the boundary lubrication regime. When the sliding velocity

increases, the pressure in the film increases and the film is able to support a larger portion of the normal load. The distance between the surfaces increases and asperities are gradually separated from each other. As a consequence, friction drops. The overall measured friction is now a combination of asperity-to-asperity friction and lubricant film friction. The velocity at which friction starts to drop depends on local (surface) parameters such as surface topology and roughness, but also global geometrical parameters such as radius of curvature of the guide and the chain links and modulus of these components. This velocity domain is termed the mixed lubrication regime.

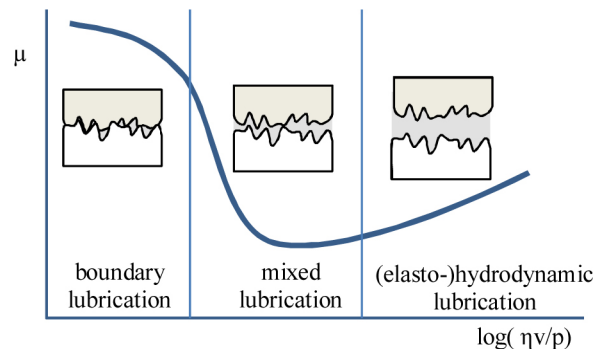


Figure 3. Schematic Stribeck curve for a sliding contact between two surfaces lubricated by a liquid. On the vertical axis is the coefficient of friction (μ). On the horizontal axis is the Hersey number, which is the product of sliding velocity (v) and viscosity (η) divided by normal pressure (p).

Further increasing the sliding velocity leads to a further increase of oil film pressure enabling it to support the full normal load of the chain and separating the surfaces completely. This condition is reached near the minimum in the Stribeck curve. At even higher velocities, the friction increases again due to the viscous drag in the oil. This velocity domain is termed the (elasto-)hydrodynamic regime. In this regime, the main friction determining parameters are the oil viscosity and the moduli of the tensioner arm, the guide, and the chain.

Friction in a real timing chain drive system is far more complex. The conditions along the chain tensioner arm and guide may vary (see Figure 4). Consider for example location 1 in this Figure. The radius of curvature of the arm is relatively large and the normal load relatively low. At sufficiently high sliding velocity, friction may be in the mixed lubrication regime at this location as indicated by the red dot in the Stribeck curve. At another point (for example location 2), the normal loading conditions may be much more severe due to the smaller radius of curvature of the tensioner arm or due to dynamics in the chain. Consequently, friction at this location may be in the boundary lubrication regime. The total friction measured over the full chain drive system is a weighed sum of the many individual contributions at all locations and this leads to a very complex relation to parameters such as chain tension, sliding velocity and temperature.

The friction losses are usually determined from the measured drive torque at one of the sprockets. These losses thus also include those due to chain articulation, chain to sprocket contact, in bearings, and at cam-follower interfaces. The interpretation of the friction losses is even

further complicated by edge effects at the chain links (stress concentrations, see Figure 10), (small) defects of the chain links due to stamping, and variations in the axial chain force and sliding velocity.

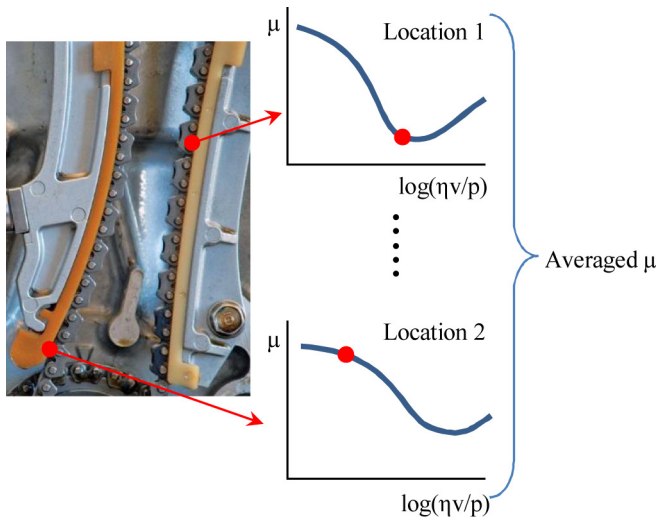


Figure 4. Schematic representation of the coefficient of friction in a chain to tensioner arm contact.

BALL-ON-PYRAMID TEST

The ball-on-pyramid test set-up (see Figure 5) is used here in support of deepening the understanding of the mechanisms of lubricated friction and for identifying the most promising routes for lowering friction by adaptation of the chain tensioner arm and guide materials.

A sketch of the testing principle is shown in Figure 6. For a single test, 3 identical samples with dimensions $4 \times 6 \times 15 \text{ mm}^3$ are placed under a 45° orientation angle as shown in the Figure. A chromium steel ball (12.7mm diameter) is placed in the center and is supported by the three samples. A normal load is applied and the ball is rotated around its vertical axis. The torque required to rotate the ball is logged and the coefficient of friction is derived from this torque. The contacts are submerged in oil, and the temperature of the oil is controlled via a heating stage.



Figure 5. Ball-on-pyramid test set-up. The device used here is the commercially available Anton Paar Tribo-cell T-BTP installed in an Anton Paar MCR 501 Rheometer [3]. The left image shows the Tribo-cell installed in the rheometer and the right image the chromium steel ball (ISO 3290 G20) supported by 3 identical small plates of the material to be tested.

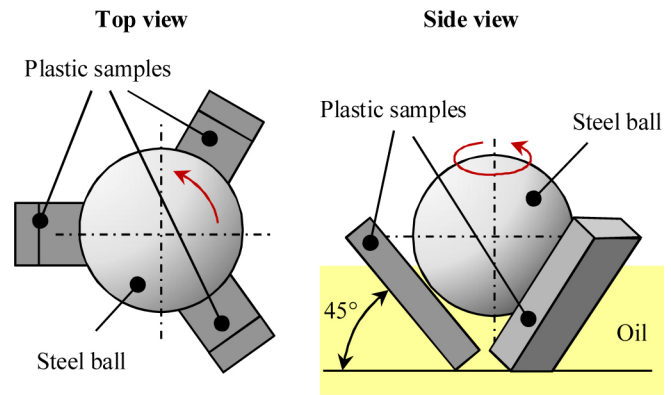


Figure 6. Schematic drawing of the ball-on-pyramid set-up. The plastic samples are placed under a 45° angle. The three contact points are submerged in oil. The diameter of the ball is 12.7mm.

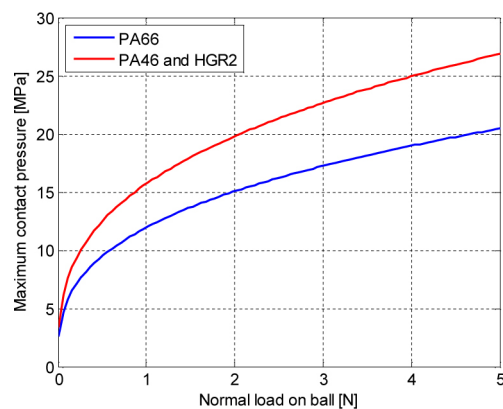


Figure 7. Estimated maximum contact pressure (Hertzian contact mechanics, see Appendix A) as a function of the normal load on the chromium steel ball for PA66, PA46, and HGR2 at 90°C . The modulus of HGR2 (PA46 with small amounts of friction modifiers) is nearly equal to that of PA46 and therefore, its estimated maximum contact pressure is equal to that of neat PA46.

The test allows for a fairly straightforward analysis and the test conditions are well-defined and well-controlled. However, the correlation between this test and a real chain tensioner arm contact is quite complex and not fully understood, so care should be taken to extrapolate results of this test to the real application. Nevertheless, it captures some of the essentials in a chain to tensioner arm or guide contact, *i.e.* converging wedge, steel and plastic surface contact in the presence of a lubricating oil.

Performing a test requires only a small amount of sample material in a simple geometry which makes it easy to screen a number of materials. The samples can be taken from molded parts or standard tensile test bars.

An important parameter in friction is the normal pressure. Using Hertzian contact mechanics, the maximum pressure at the three contact points is estimated as a function of normal load on the ball (see Figure 7) for PA66 and PA46 at 90°C . Note that the maximum pressure is dependent on the modulus of the tested material and consequently varies with material type and temperature (see Table 1 for modulus values used in the calculation). The standard normal load

used in the tests described in this paper is 1N. For this value, the maximum contact pressure is nearly 12 MPa for PA66 and nearly 16 MPa for PA46.

Table 1. Modulus (DAM) of PA66 and PA46 at several temperatures. The modulus of HGR2 is nearly the same as that of PA46.

Temperature	PA 66	PA46
20°C	3.4 GPa	3.3 GPa
90°C	0.7 GPa	1.0 GPa
130°C	0.5 GPa	0.7 GPa

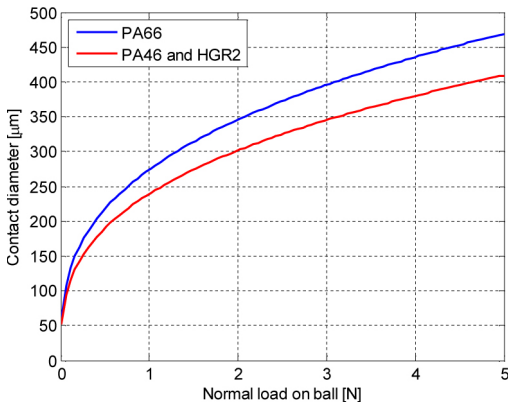


Figure 8. Estimated contact diameter (Hertzian contact mechanics, see Appendix A) as a function of normal load on the steel ball for PA66, PA46, and HGR2 at 90°C. The modulus of HGR2 (PA46 with small amounts of friction modifiers) is nearly equal to that of PA46 and therefore, its estimated maximum contact pressure is equal to that of neat PA46.

The calculated diameter of the contact spot is shown in Figure 8. At 1N normal load the estimated contact diameter is roughly 230µm to 270µm (depending on the type of plastic material).

The test set-up does not allow for real-time measurement of the contact diameter under the normal load. Figure 9 shows the wear mark on a Stanyl HGR2 sample (PA46 with small amounts of friction modifiers) after a friction test (normal load 1N, 90°C oil temperature, Castrol Edge 5W30 engine oil). The diameter of the wear mark is estimated at 250µm to 300µm which is in good agreement with the calculated values.

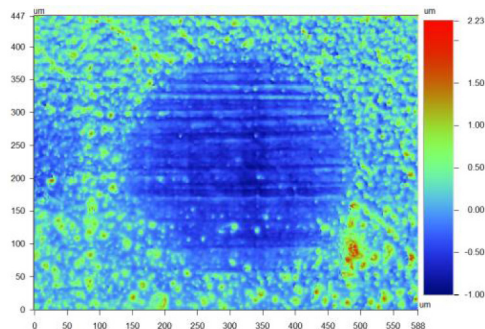


Figure 9. Wear mark after ball-on-pyramid friction test for a Stanyl HGR2 sample (PA46 with small amounts of friction modifiers). The test conditions were: normal load 1N, oil temperature 90°C, Castrol Edge 5W30 lubricant. Sliding of the ball was in horizontal direction. The image was generated using optical profiler (Wyko NT1100).

COMPARISON OF CHAIN-ON-GUIDE AND BALL-ON-PYRAMID

Although the ball-on-pyramid test is of different geometry (point contact) than an actual chain on guide set-up (line contact), it captures a many of the relevant characteristics of friction under oil lubrication. The main conditions relevant for friction of the two setups are compared in Table 2.

Table 2. Comparison of chain-on-guide set-up and ball-on pyramid set-up.

	Chain-on-guide	Ball-on-pyramid
Contact	Line contact	Point contact
Radius curved surface	100 – 1000mm	~6.3 mm
Temperature	80°C – 140°C	25°C – 150°C
Contact pressure	1MPa – 20MPa	10MPa – 20MPa
Sliding velocity	1m/s – 15m/s	10 ⁻⁴ m/s – 1.5m/s
Material	Chromium steel	Chromium steel
Modulus at 25°C	~200 GPa	~200 GPa
Surface roughness	R _A ~ 0.2 – 0.4µm	R _A ~ 0.1 µm

Generally, the conditions of both set-ups are quite similar except for the sliding velocity. The maximum attainable sliding velocities for the ball-on-pyramid set-up is on the low boundary of those encountered in an actual chain-on-guide contact.

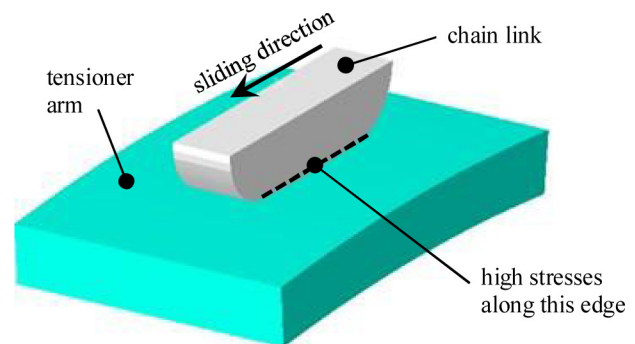


Figure 10. Finite element model used to calculate the stresses in the contact between a chain link and tensioner arm. The dashed line marks one of the chain link edges along which stress concentrations are expected.

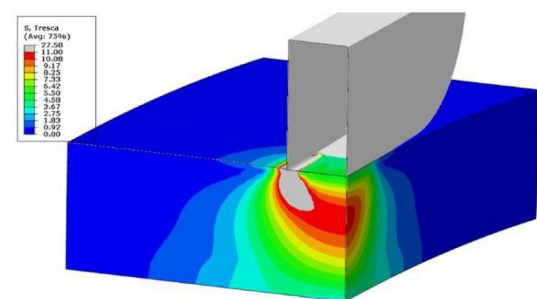


Figure 11. Example of calculated stress distribution (equivalent Tresca stress) in the tensioner arm near the contact with the chain link. The stress distribution is shown on a quarter section of the model in Figure 10. Calculated peak stress is on the order of 27 MPa.